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COLD-AIR INVESTIGATION OF PROTOTYPE SNAP-8 TURBINE

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SUMMARY

A two-stage prototype turbine that was designed and built by the Aerojet-General Corporation for the SNAP-8 system was investigated experimentally in cold air over a range of blade- to jet-speed ratios from 0.13 to the design value of 0.3. A total-to-static efficiency of 0.62 was obtained at the design blade- to jet-speed ratio. This value of efficiency is 0.034 less than the design value of 0.654. An additional 0.015 drop in efficiency was obtained when the efficiency was based on the static pressure measured at the exit of the diffuser.

The results of the investigation also showed an equivalent weight flow of 0.0994 pound per second at the design blade- to jet-speed ratio compared with the design value of 0.083. This large variation from the design weight flow was attributed to the fact that the first-stage stator throat area was about 30 percent larger than the design value. Because of the mismatch in areas throughout the turbine, design pressure distribution through the turbine was not obtained. This resulted in a ratio of work output of the first stage to that of the second stage of 1 to 2.1; the design ratio was 1 to 1.4.

INTRODUCTION

As part of a general NASA program to develop electrical power generating systems for space applications, the Aerojet-General Corporation is under contract for the development of the SNAP-8 turboelectric system. The system employs a Rankine cycle with mercury as the working fluid and a turbogenerator as the unit for conversion of nuclear power to electrical power.

In order to facilitate the development program by providing turbine performance information to the contractor and to learn whether the unusual design features of the turbine would penalize its performance level, a prototype turbine supplied by the contractor was evaluated at the Lewis Research Center. The turbine is a two-stage full-admission unit with values of design air equivalent weight flow and blade- to jet-speed ratio of 0.083 pound per second and 0.3, respectively. The design work split is 1 to 1.4. Unusual features of this turbine are the stator blade shapes and the large difference between the mean diameters of the first and second stages.

This report presents the results of a cold-air evaluation of the subject turbine. The results include the overall performance obtained over a range of blade- to jet-speed ratios from 0.13 to the design value of 0.3.

SYMBOLS

g gravitational constant, 32.17 ft/sec^2

Δh specific work output, Btu/lb

J mechanical equivalent of heat, 778.16 ft-lb/Btu

M Mach number

N rotative speed, rpm

p absolute pressure, lb/sq ft

r radius, in.

U blade velocity, ft/sec

V absolute gas velocity, ft/sec

W relative gas velocity, ft/sec

w weight-flow rate, lb/sec

γ ratio of specific heats

δ ratio of turbine-inlet total pressure to that of NACA standard sea-level atmosphere, $p_1'/2116$

ϵ function of γ used in relating weight flow to that using inlet conditions at NACA standard sea-level atmosphere,

$$\frac{0.740}{\gamma}\left(\frac{\gamma + 1}{2}\right)^{\frac{\gamma}{\gamma - 1}}$$

η adiabatic efficiency, ratio of blade power to ideal blade power based on inlet-total- to exit-static-pressure ratio

θ_{cr} squared ratio of critical velocity of turbine inlet to critical velocity of NACA standard sea-level atmosphere, $(V_{cr}/1019)^2$

ν blade- to jet-speed ratio, $U_m/\sqrt{2gJ\ \Delta h_{id}}$

Subscripts:

cr conditions at Mach number of unity

h hub

id ideal

m mean radius

t tip

x axial direction

1 station at turbine inlet

2 station at outlet of first-stage stator

3 station at outlet of first-stage rotor

4 station at outlet of second-stage stator

5 station at outlet of second-stage rotor

6 station at exit of diffuser

Superscript:

' absolute total state

TURBINE DESCRIPTION

The following design requirements for the subject turbine using mercury vapor as the working fluid were supplied by the contractor:

Overall total-to-static efficiency, η_{1-5} 0.654

Overall total- to static-pressure ratio, p_1'/p_5 . . . . 14.47

Rotative speed, N, rpm 20,000
Inlet temperature, $^{\circ}F$ 1200
Inlet pressure, lb/sq in. abs 275
Overall blade- to jet-speed ratio, ν 0.3
First-stage blade- to jet-speed ratio 0.425
Second-stage blade- to jet-speed ratio 0.425
Work split (ratio of work output of first stage to that of
second stage) 1:1.4

A design value of air equivalent weight flow $(w\sqrt{\theta_{cr}}/\delta_1)\epsilon$ of 0.083 pound per second was calculated from the design turbine-inlet conditions for a choked first-stage stator with a total throat area of 0.2495 square inch (as stated by the contractor) and an assumed flow coefficient of 0.97.

Velocity Diagrams

The velocity diagrams furnished by the contractor are given in figure 1 for turbine operation in mercury vapor at an inlet temperature of 1200° F. The diagrams were computed with the assumption that the total- to static-pressure ratio across the stage was taken across the stator, the first- and second-stage stator pressure ratios being 2.64 and 5.47, respectively. The two stators were, thus, designed to operate in a choked condition. The exit angle, measured from the tangential direction, is 17° for both stators. There is a small decrease in relative Mach number through the first-stage rotor, the inlet and outlet relative Mach numbers being 0.660 and 0.554, respectively. The relative Mach number also decreases as the flow moves through the second-stage rotor; the relative Mach number is 0.936 at the inlet and 0.805 at the outlet. Both rotors are designed for a turning angle of 118°. The diagrams in figure 1 also show some positive exit whirl out of each stage.

Stators

The stator- and rotor-blade profiles and flow passages are shown in figure 2. The first-stage stator assembly contains 30 blades, which are merely straight vanes 0.030-inch thick that were machined at an angle of 17° from the tangential direction. The inlet housing is designed to impart a tangential velocity to the fluid in order to minimize entrance losses. The outer boundary of the flow passage is a cylindrical surface 4.170 inches in diameter. The inner boundary of the passage is contoured from stator inlet to stator exit to give, initially, a rapidly decreasing flow area and, then, from the throat to the trailing edge, an increasing flow area. The blade height is 0.212 inch at the leading edge, 0.089 inch at the throat, and 0.110 inch at the trailing edge. The solidity of this stator, based on the spacing at the trailing edge at the mean radius, is 2.01.

A flow-area ratio of 2.33 between the first- and second-stage stator throat areas is obtained partly by an increase in the annulus mean diameter immediately downstream of the first-stage rotor and partly by an increase in the blade height at the throat. As a result the second-stage stators have a maximum tip diameter of 4.96 inches compared with 4.17 inches for the first-stage stators. The second-stage stator assembly contains 31 blades. The desired variation in flow area through the stator is obtained by the blade profile and by the contour of the inner and outer surfaces of the passage. For a short distance from the leading edge of the blade the flow area is greatly affected by an extremely large nose radius of about 0.08 inch. The flow area increases sharply from the throat to the trailing edge of the blade because of the contour of the inner surface. This yields an area ratio of 1.61. The blade height is 0.220 inch at the leading edge, 0.175 inch at the throat, and 0.282 inch at the trailing edge. The solidity of this stator, (based on the spacing at the trailing edge at the mean radius) is about 2.5.

Rotors

The turbine rotors are shown in figure 3. The first-stage rotor assembly has a mean diameter of 4.06 inches and contains 79 blades with a height of 0.103 inch. The second-stage rotor assembly has a mean diameter of 4.65 inches and contains 91 blades with a height of 0.285 inch. The aspect ratios, based on the blade chord, are 0.367 and 1.017 for the first and second stages, respectively. The solidities are 1.74 for both blade rows. Both rotors were designed for constant static pressure from inlet to exit, with blades of circular arc design and a gradually increasing width of channel from inlet to exit to account for losses. The blades were machined integrally with the disk.

Diffuser

The diffuser section has an inlet flow area of 4.65 square inches and an outlet flow area of 9.62 square inches. It was designed to turn the flow through an angle of about 90° immediately downstream of the second-stage rotor and then to recover some of the velocity head before discharging the flow into the exhaust system.

APPARATUS, INSTRUMENTATION, AND PROCEDURE

The apparatus used in the performance evaluation of the subject turbine consisted of an airbrake and piping into the laboratory air and exhaust system. Figure 4 shows a photograph of the experimental turbine installation. High-pressure dry air was supplied from the laboratory air system at an average temperature of 580° R. The air was filtered to remove dirt particles before it entered the turbine through the spiral-shaped inlet housing. After passing through the turbine, the air was exhausted into the laboratory low-pressure exhaust system. A remotely controlled pressure-regulating valve was used on the high-pressure air to throttle it and to maintain the inlet pressure constant. Remotely controlled valves in the low-pressure exhaust system were used to maintain the desired pressure ratio across the turbine. Flow to the airbrake was supplied from the laboratory air supply system at approximately 40 pounds per square inch gage. After the air was throttled to the desired pressure, it entered the inlet manifold, passed through the brake, and was discharged axially into the test cell.

A cutaway drawing of the turbine test section and the airbrake power absorption assembly is shown in figure 5. As indicated in the drawing, the airflow enters the turbine inlet, passes through the two stages of blading, and is discharged through the diffuser section and the outlet pipe. The drawing also shows the direction of flow through the airbrake. In order to obtain an accurate power measurement, air is brought into the brake from the inlet manifold in an axial direction and is discharged in the same direction; this makes the torque of the rotor equal to that on the airbrake casing, which is cradle-mounted on air bearings. A description of a similar airbrake is presented in reference 1.

Instrumentation was provided in the turbine test section in order to obtain overall performance and interstage temperatures at various speeds and pressure ratios. The airflow was measured by a calibrated orifice plate. The torque output of the turbine was measured with a calibrated commercial strain-gage load cell. The rotational speed of the turbine was measured with an electronic counter in conjunction with a magnetic pickup and a shaft-mounted gear. All pressure measurements were made from the various pressure taps, shown in figure 2, by calibrated electrical pressure transducers. The turbine-inlet total pressure (station 1) was measured with two static-pressure taps, which were located in the inlet housing about 180° apart at points of low Mach number, where the total- to static-pressure ratio can be assumed equal to 1.0. Turbine overall performance was based on an average of the static pressures measured with seven static-pressure taps located immediately downstream of the second-stage rotor trailing edge (station 5). The taps were spaced around the annulus with three on the inner wall and four on the outer wall. The performance of the turbine and diffuser was based on the static pressure measured with the tap located at the exit of the diffuser (station 6). Temperatures were measured with thermocouples at the inlet to the turbine (station 1), at the exit of the first-stage rotor (station 3), and at the diffuser exit (station 6). All data were recorded by an automatic digital potentiometer and were processed through an electronic computer.

Experimental data were taken over a range of inlet-total- to exit-static-pressure ratios from 8 to 14. At each pressure ratio, the turbine rotor speed was varied over a range of speeds from about 11,750 to about 27,800 rpm. The average turbine-inlet temperature was about 580° R, and the inlet total pressure was approximately 35 pounds per square inch absolute.

The friction torque of the bearings and the seals was obtained by motoring only the shaft at various speeds and measuring the torque with a strain-gage load cell. Then the power calculated from this torque was added to the shaft power to obtain the turbine rotor power. The turbine efficiency was computed as the ratio of actual rotor output to ideal output, as calculated from the weight flow, inlet total temperature and pressure, and outlet static pressure.

RESULTS AND DISCUSSION

The performance characteristics of the subject turbine are presented in figures 6 and 7. Figure 6 shows the variation of total-to-static efficiency η_{1-5} with blade- to jet-speed ratios from 0.13 to the design value of 0.3. The efficiency and jet-speed calculations for this figure are based on the average static pressure measured immediately downstream of the second-stage rotor trailing edge (station 5). Thus, these calculations do not include the performance of the diffuser. The data points describe a curve that is similar to those for conventional, two-stage turbines (refs. 1 and 2). The curve shows a value of total-to-static efficiency of 0.62 at the design value of blade- to jet-speed ratio of 0.3. This efficiency is 0.034 less than the design value of 0.654. Therefore, the performance level of the turbine was not greatly penalized by the unusual design features.

In order to obtain the overall performance of the turbine with the diffuser, efficiency was calculated on the basis of the static pressure measured at the outlet of the diffuser (station 6). Figure 7 shows the results of these calculations as the variation of total-to-static efficiency for the turbine and the diffuser η_{1-6} with blade- to jet-speed ratio ν_{1-5}. (The blade- to jet-speed ratio is based on the average static pressure at station 5.) The resulting curve is similar to that of figure 6, which is shown here as a dotted line; however, all values of efficiency for the turbine and diffuser are lower than those for the turbine at the same value of blade- to jet-speed ratio. At design blade- to jet-speed ratio the curve for the turbine and diffuser shows a value of 0.605, which is 0.015 lower than that for the turbine. Thus, instead of showing an increase in efficiency due to recovery of some velocity head in the diffuser, the results actually show a small decrease in efficiency. The failure of the diffuser to recover this velocity head may be due to the excessive losses caused by flow separation at the sharp turn immediately downstream of the second-stage rotor.

A value of equivalent weight flow $(w\sqrt{\theta_{cr}}/\delta)\epsilon$ of 0.0994 pound per second was obtained at design blade- to jet-speed ratio. This value of weight flow is about 20 percent larger than the design value of 0.083. Since the turbine was designed to operate with a choked first-stage stator, the results indicate a value of first-stage stator throat area considerably larger than the design value. In order to obtain more definite information as to the first-stage stator throat area, a choked-flow check was made of the stator. The results of this check showed a choking equivalent weight flow of 0.108 pound per second. Thus, the first-stage stator operated in the unchoked condition with a throat area about 30 percent larger than the design value.

The work split between the two stages was also effected by the mismatch in the flow areas. With a two-stage turbine operating at a given overall pressure ratio, an increase in first-stage stator throat area would decrease the pressure ratio across the first stage and increase that across the second stage. Since the work output is a function of the pressure ratio, it would then be expected that the design work split would not be obtained. The actual value of this split was calculated from the measured temperature distribution through the turbine. The work split between the two stages was such that the second-stage work output was about 2.1 times that of the first stage, which is compared with the design second-stage output of 1.4 times that of the first stage.

SUMMARY OF RESULTS

An experimental cold-air investigation of the SNAP-8 prototype two-stage turbine yielded the following results:

1. A total-to-static efficiency of 0.62 was obtained at the design blade- to jet-speed ratio of 0.3; calculations were based on the average static pressure measured immediately downstream of the second-stage rotor trailing edge. This efficiency is 0.034 less than the design value of 0.654, which indicates that the turbine performance was not greatly penalized by the unusual design features.

2. A total-to-static efficiency of 0.605 was obtained for the subjet turbine with the exit diffuser at the design blade- to jet-speed ratio, the efficiency calculation being based on the static pressure measured at the exit of the diffuser. The failure of the diffuser to

recover the velocity head at the turbine exit was probably due to the excessive losses caused by flow separation at the sharp turn downstream of the second-stage rotor.

3. A value of equivalent weight flow of 0.0994 pound per second was obtained at the design blade- to jet-speed ratio. This value is about 20 percent larger than the design value of 0.083 pound per second. A choked-weight-flow check of the first-stage stator indicated that the first-stage stator throat area was about 30 percent larger than the design value.

4. The ratio of the work output of the first stage to that of the second stage was about 1 to 2.1 compared with the design ratio of 1 to 1.4. The failure to obtain the design work split was attributed mainly to the large first-stage stator throat area with the resultant change in pressure distribution throughout the turbine.

Lewis Research Center
National Aeronautics and Space Administration
Cleveland, Ohio, July 23, 1962

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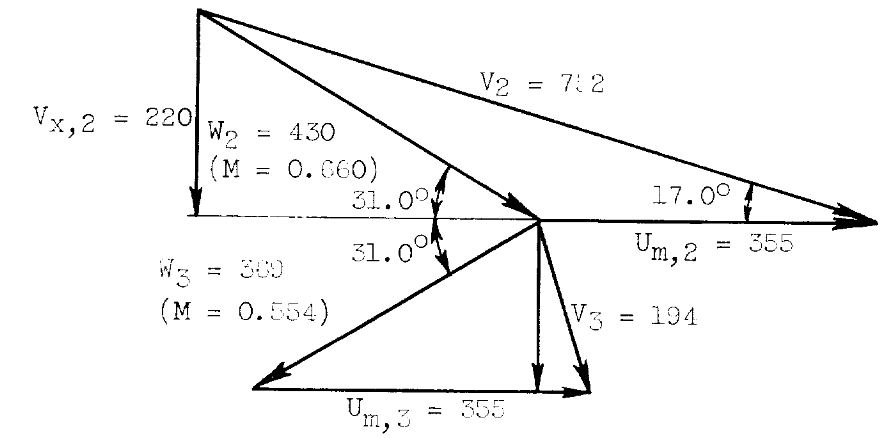
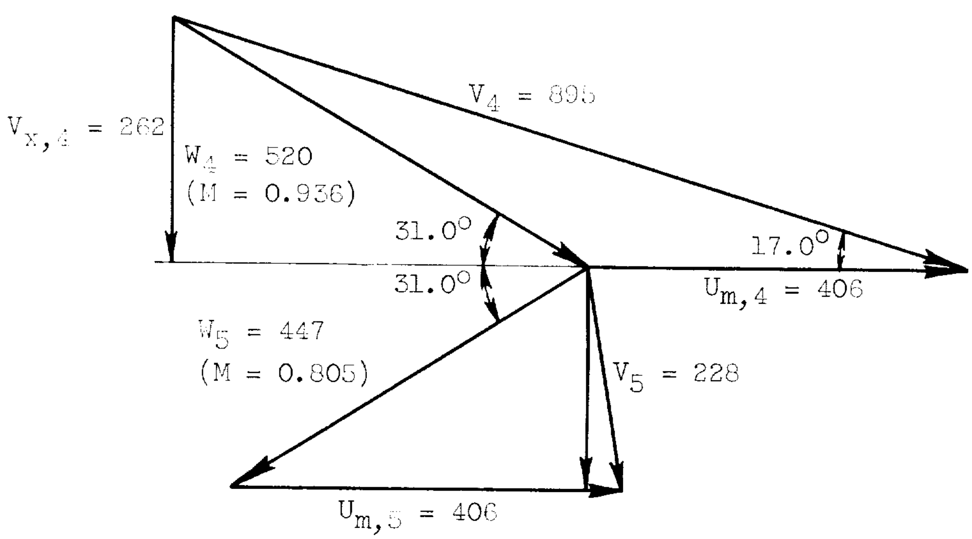
2. Wong, Robert Y., and Darmstadt, David L.: Comparison of Experimentally Obtained Performance of Two Single-Stage Turbines with Design Ratios of Blade to Jet Speed of 0.191 and 0.262 Operated in Hydrogen and in Nitrogen. NASA TM X-415, 1961.



(a) First stage.



(b) Second stage.

Figure 1. - Free-stream velocity diagrams. (Velocities are in ft/sec.) Working fluid, mercury vapor; inlet temperature, 1200° F.

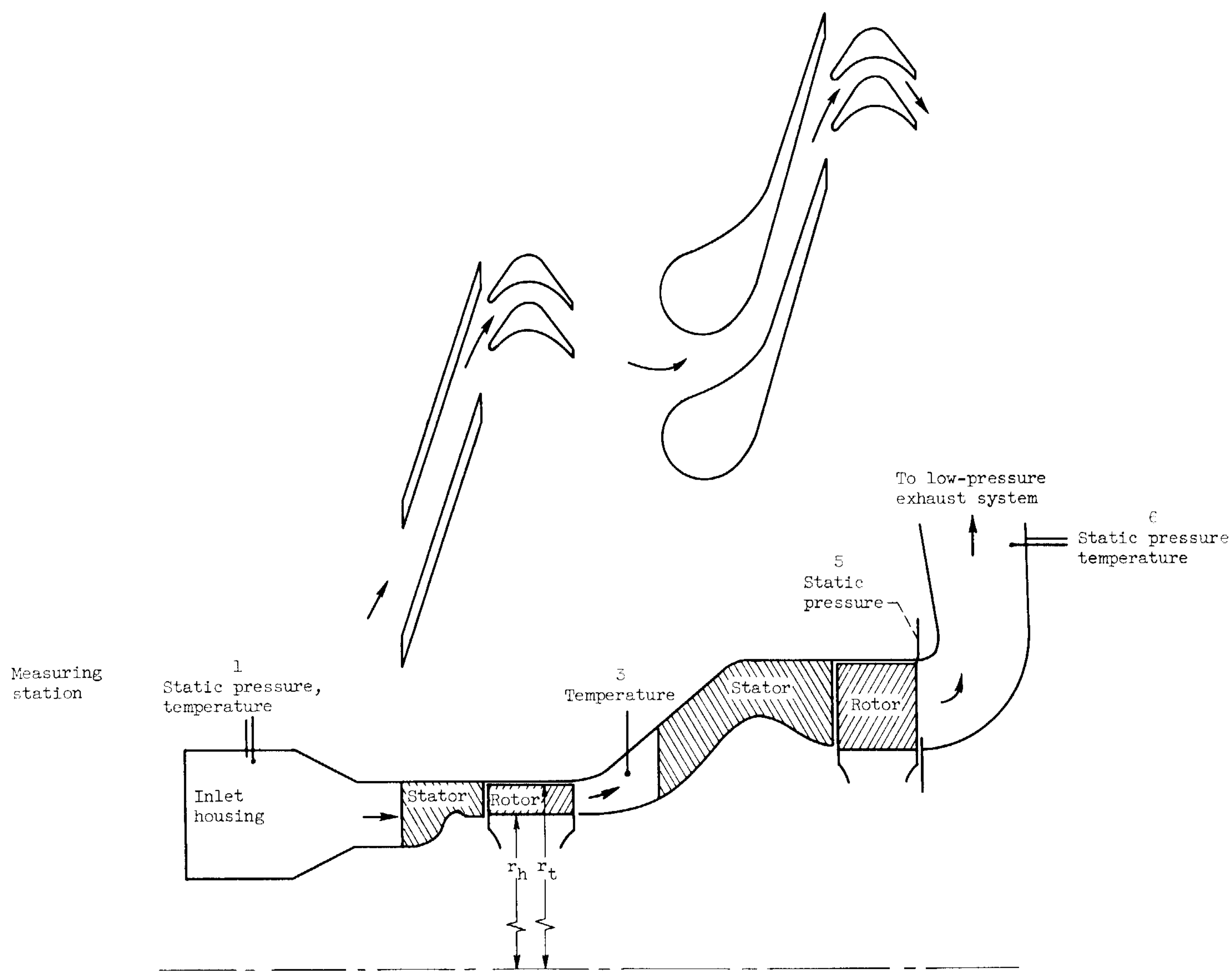


Figure 2. - Stator- and rotor-blade profiles and flow passage.

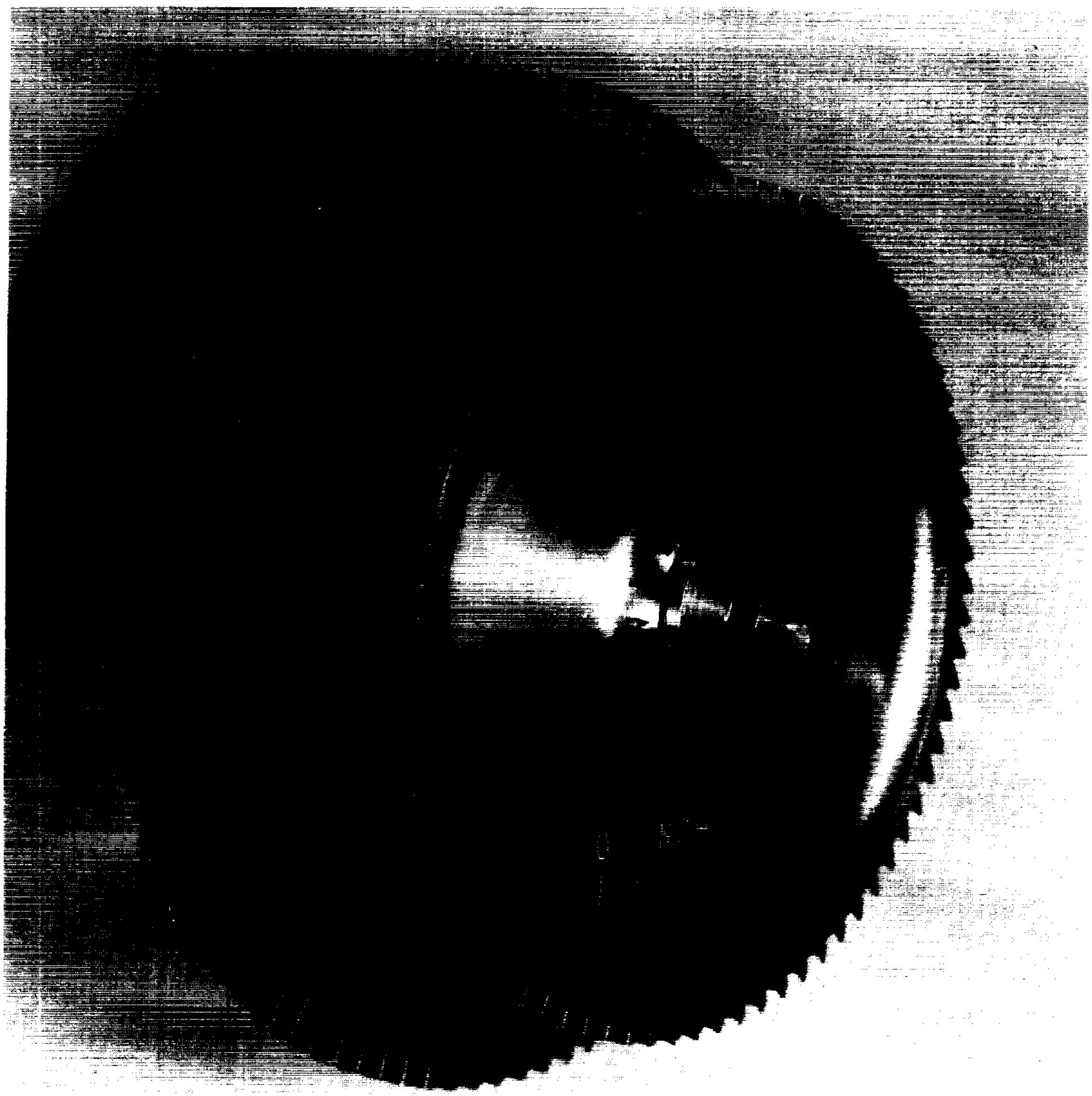


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Figure 3. - Turbine rotors.

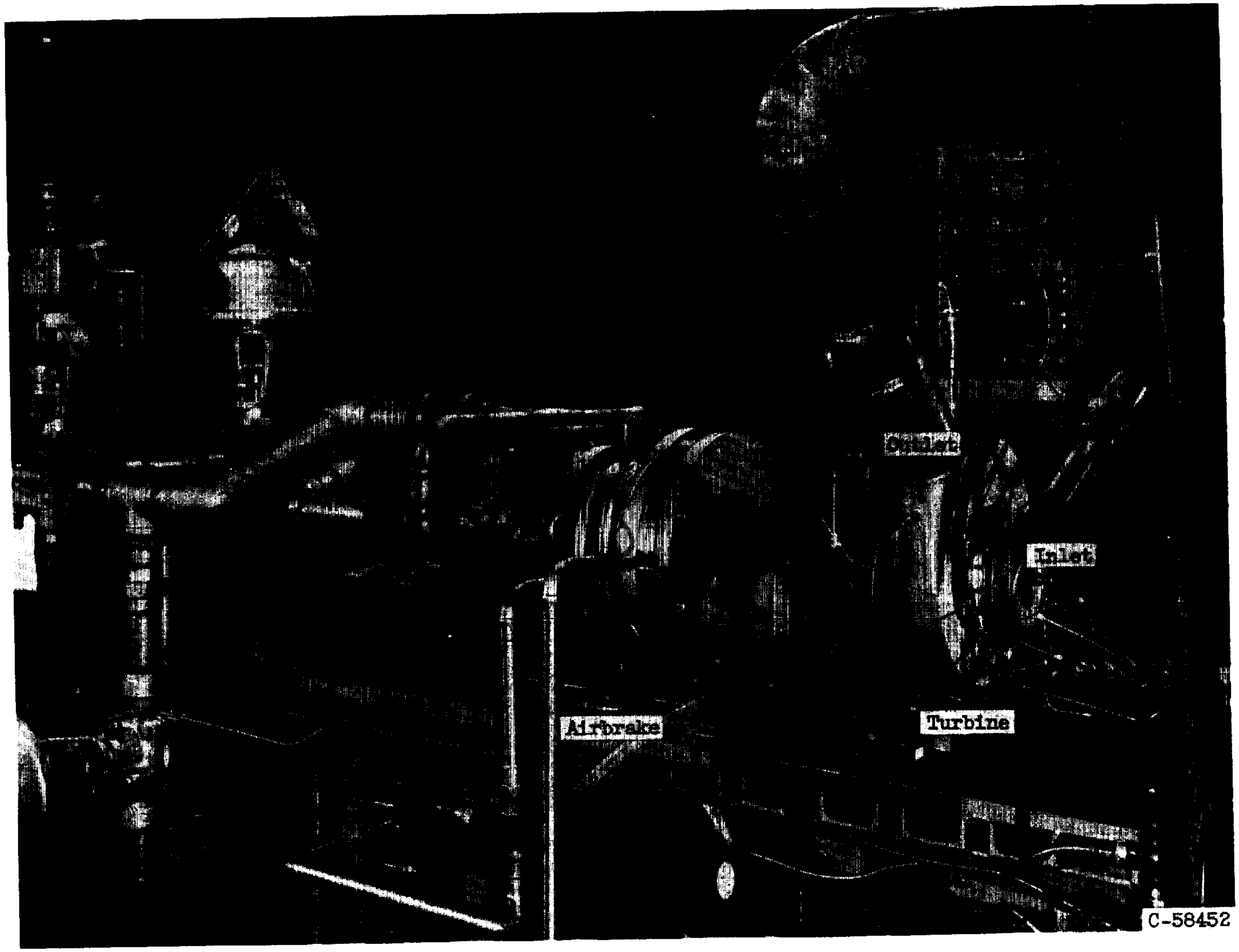

Figure 4. - Turbine test apparatus.

Figure 5. - Cutaway view of turbine and airbrake assembly.

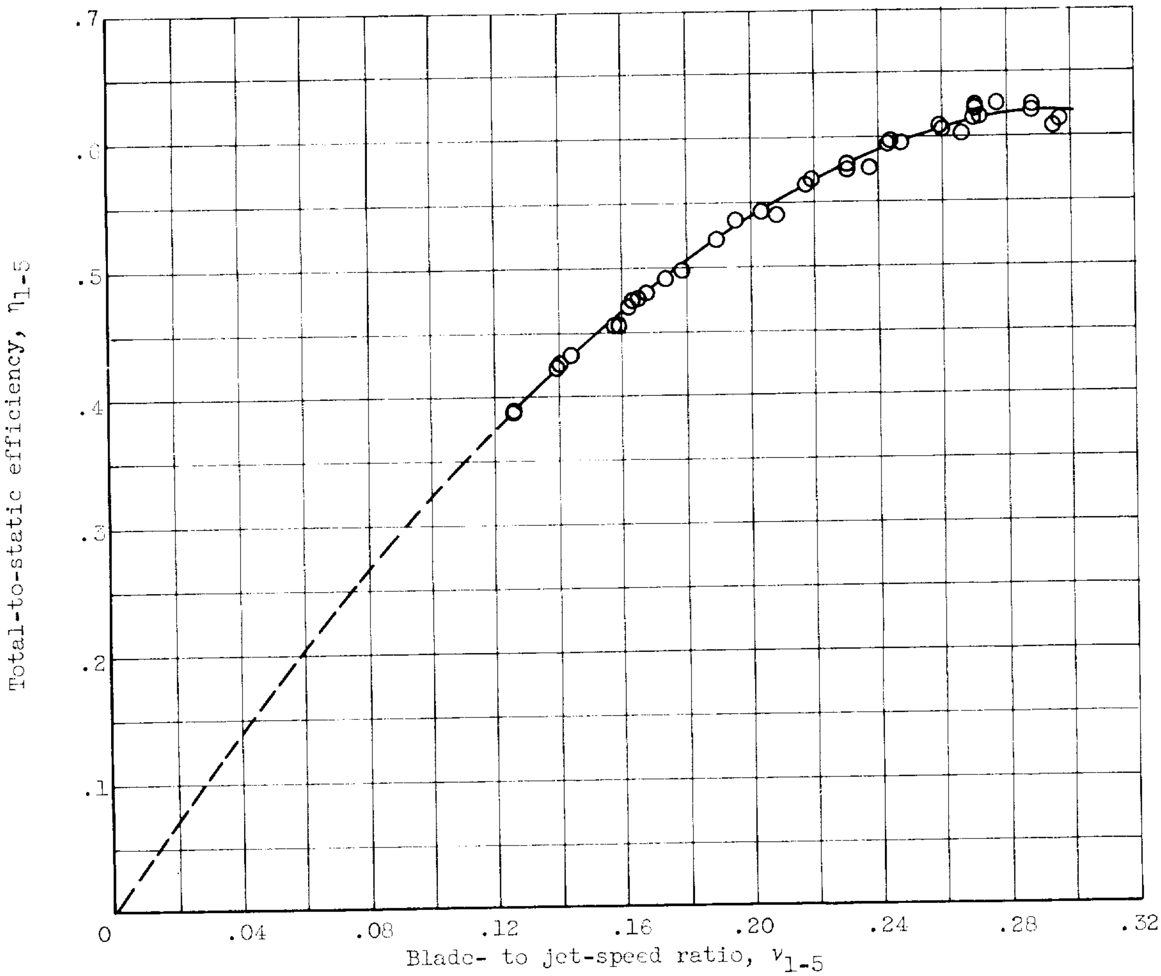

Figure 6. - Variation of efficiency based on inlet total and average turbine-outlet static pressure with blade- to jet-speed ratio.

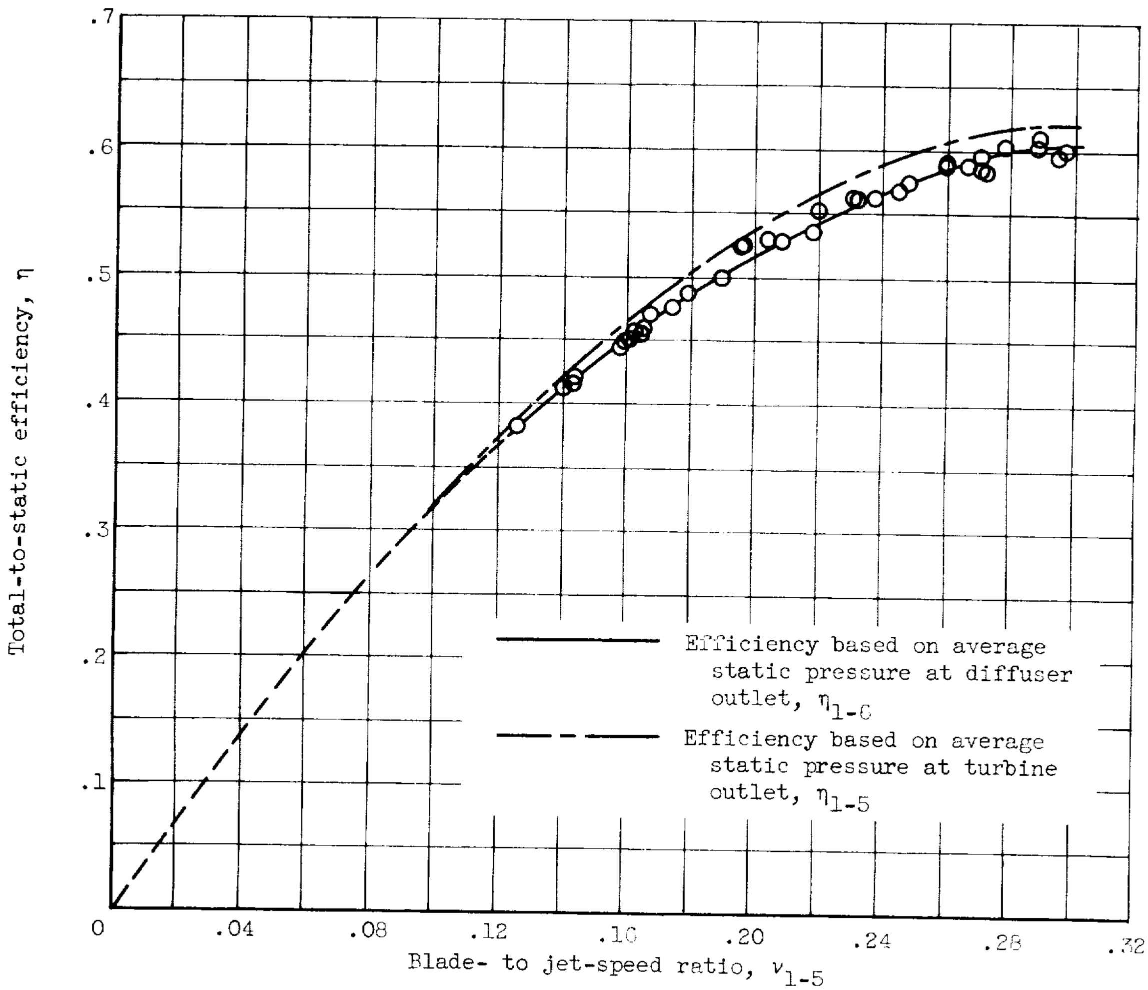

Figure 7. - Variation of efficiency based on inlet-total and diffuser-outlet-static pressure with blade- to jet-speed ratio.